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By Herbert J. Venediger

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EVALUATION OF SCAVENGING IN TWO-STROKE-CYCLE ENGINES*

By Herbert J. Venediger

Previous researches on the scavenging of two-stroke-cycle engines served to determine the following evaluation coefficients:

1. The quantitative scavenging efficiency η_s;

2. The qualitative scavenging efficiency η_s';

3. The two-dimensional scavenging-type efficiency η_{sf};

4. Utilization of the energy of the scavenging medium η_A.

Chronologically considered, the order of succession was reversed, the start being originally from the scavenging pattern (represented by 3). Thus the individual scavenging methods (uniflow scavenging, cross scavenging, counterflow scavenging, loop scavenging, and turbulent scavenging) and their combinations were developed. The flow pattern was generally obtained in a simple manner and only with rough approximation by blowing smoke into a glass cylinder model or by scavenging the latter at low velocity with air containing pyrotechnic bodies, sawdust, or the like.

Such methods have now been abandoned. For some years special attention has been given to qualitative evaluation with the engine running, and the methods of testing have been so improved that the analysis of the exhaust gases for their content in CO and CO_2 can be made with great accuracy by means of extremely small controlled valves, even at high revolution speeds ($n = 2500$ r.p.m.) (reference 1). If V_s = amount of scavenging medium introduced into cylinder,

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*"Wertung der Spülung von Zweitaktmotoren." Automobiltechnische Zeitschrift, June 25, 1933, pp. 301-308.

V_n = permanent useful portion,

V_r = residual gas in m³ (p_o, T_o), we then have

$$\eta_s' = \frac{V_n}{V_n + V_r} \qquad (1)$$

for the qualitative evaluation.

For the quantitative evaluation which has thus far been determined only from model tests, we have

$$\eta_s = \frac{V_n}{V_s} \qquad (2)$$

This efficiency coefficient accordingly yields information regarding losses of the scavenging medium and is of special importance in engines charged with gas mixtures. The two evaluations are related through the independent quantity $\alpha = V_r/V_s$ in the formula

$$\eta_s = \alpha \frac{\eta_s'}{1 - \eta_s'} \qquad (3)$$

$$\eta_s' = \frac{\eta_s}{\eta_s + \alpha} \qquad (4)$$

Equation (3) means that, for η_s' (i.e., complete scavenging), the amount of the scavenging medium must be infinitely great, a fact already established by Neumann with the aid of a complicated formula. A rough approximation can be quickly made with equations (3) and (4). If, for example, $\alpha = 0.40$ (a value observed in two-stroke-cycle engines with crankcase scavenging), it follows that, for $\eta_s = 0.70$ (a value found in cross-scavenging), no higher qualitative scavenging efficiency than $\eta_s' = 0.63$ can be attained. For $\eta_s' = 0.72$, η_s would have to be 1.0, i.e., there could be no losses of the scavenging medium.

If, on the other hand, the qualitative evaluation of the scavenging of such engines at a given revolution speed yields the best value $\eta_s' = 0.82$, we then obtain an α value of 0.175 with $\eta_s = 0.70$, while Neumann (reference 2) with a Junkers opposed-piston engine, with twice as large charging pump and turbulent scavenging, found an α value of 0.05.

The Two-Dimensional Scavenging-Type Efficiency

The flow in the cylinder cannot be closely followed mathematically. One is thus compelled to plot the form of the scavenging curve according to the general laws of the flow theory in the cylinder and usually assumes that the flow is practically steady. A certain mean flow direction and pressure are then coordinated for every point in the cylinder.

Hitherto the scavenging pattern has always been determined simply for the largest longitudinal cylinder section, which is termed the main longitudinal cylinder section. In this section the scavenging pattern is known to show, for every scavenging method, incompletely scavenged cylinder zones, the size and variation of which are to be determined. These zones are called head vortices near the cylinder head and piston pockets near the piston head. The location of these zones can be determined quite accurately (their variable size during the scavenging, less accurately) by drawing the scavenging pattern.

If the scavenging pattern is drawn, two fundamentally possible forms are first obtained for all scavenging methods in which the exhaust and scavenging ports are at the same end of the piston travel. Figures 1 and 3 represent the serviceable forms in cross and loop scavenging, while figures 2 and 4 show the deficient forms, when the scavenging and exhaust ports are short-circuited. The latter case will be discussed farther on. Both cases are stable, i.e., changes are produced only by external influences.

If the area of the main longitudinal cylinder section (including the area of the section of the combustion chamber) is designated by f, and the scavenged area (as indicated by the streamlines) by ρ, the factor $\eta_{sf} = \rho/f$ then represents the two-dimensional scavenging-type efficiency of this cylinder section. This will be the next subject discussed, insofar as not otherwise expressly stated. The exact value of ρ can be determined only under simplified assumptions based on model tests, as recently done by Lutz in his fundamental work "Untersuchungen über die Spülung von Zweitaktmotoren" (Dissertation, 1932, published by Konrad Wittwer, Stuttgart) for the loop-scavenging fundamental form I (M.A.N.). It can be shown, however, that sufficiently accurate values for ρ, at least for the maximum opening of the ports, can be determined by sketching the imaginary flow. This is important, because the efficiency of the various scavenging methods can thus be compared and a series of important conclusions based on this assumption as, for example, all the pertinent patent methods and rights.

For obvious reasons η_{sf} is intimately dependent on f, that is, on the stroke ratio of the engine and on the height of the scavenging port. Experiments have shown that η_{sf} reaches its maximum at a very definite height of the scavenging port, and that therefore η_{sf} does not continue to increase in proportion to the height of the port (as Lutz concluded in the above-mentioned dissertation, page 48). For obvious reasons the development of the desired type of flow is physically impossible beyond a certain height of the scavenging port.

Engines of the same type and size, with scavenging ports 15 or 16 mm (0.59 or 0.63 in.) high, show, on the test stand, clearly differing behaviors, although the variation is hardly 13.5 percent. Of course this variation cannot be explained by the difference in the scavenging type alone, since a series of other influences are simultaneously involved. It is quite obvious, however, that increasing the height of the scavenging port beyond a certain limit impairs the engine in almost every respect. Hence, several viewpoints must be considered in designing the scavenging ports.

From the viewpoint of the flow, the size of the main longitudinal cylinder section (e.g., in loop and cross scavenging) is indicated as soon as the bore and stroke are settled. The quantitative dimensioning to permit the passage of the right amount of the scavenging medium is then simply a function of the width of the scavenging port. The scavenging pattern for the maximum scavenging-type efficiency $\eta_{sf\ max}$ with respect to time, i.e., with scavenging port fully open, shows, in agreement with the measurements, that the width of the stream through the outline of the longitudinal cylinder section having the maximum flow velocity cannot exceed a third of the cylinder diameter, if the form of the scavenging pattern is to be like figure 1 or 3.

The size and arrangement of the scavenging ports must therefore be considered from the following four principal viewpoints.

1. According to the form of the scavenging pattern (η_{sf}).

2. According to the size of the scavenging time cross section $\int f_s \, dt$.

3. According to the available scavenging time t.

4. According to the required engine characteristics (full speed, cruising speed, torque, best fuel consumption).

5. Turbulence of the scavenging medium, as in heavy-oil engines.

As regards item 1, various investigations have shown that the form of the scavenging pattern corresponds, on the whole, to the existing conceptions. The pattern continually changes, however, during the scavenging. With the scavenging port wide open, η_{sf} reaches its maximum in point of time, the head vortices and the piston pocket are the smallest, and the width of the core of the scavenging stream is the greatest. Conditions change in the reverse order, as the scavenging-port opening grows smaller, i.e., the scavenging pattern "breathes."

For the loop scavenging, fundamental form I, Lutz found a two-dimensional scavenging-type efficiency η_{sf} = 0.7 with wide-open scavenging port, 0.47 with port one third closed, and 0.32 with port two thirds closed. The cylinder diameter D was 250 mm (9.84 in.); the piston stroke s, 479.5 mm (18.88 in.); and the height of the scavenging port l_s, 38 mm (1.50 in.). Hence the coefficient $\beta = l_s/D$ (the "scavenging-type coefficient") has the value 0.152. This value, which is too small from the purely scavenging viewpoint, was adopted because the useful stroke (which is only 76 percent, due to the partition 37 mm (1.46 in.) high between the exhaust and scavenging ports) would obviously be too greatly reduced with better scavenging ports.

The maximum value of η_{sf} with respect to time determines the success of the scavenging. From this follows the basic scavenging requirement that the scavenging pressure with fully open port must not already have begun to decrease, but must show its permissible or requisite maximum value. This requirement determines the timing of the scavenging delivery. In this connection it must be remembered that an exhaust pressure p_z of 2 to 2.5 atmospheres exists in all high-speed two-stroke-cycle engines at the instant of opening the scavenging port. The scavenging can first begin at about 20 to 30° before the bottom dead center, according to the height of the requisite scavenging pressure p_s (reference 3). This is very favorable from the viewpoint of the scavenging type, because the exhaust gases are immediately embraced in the broad stream. In a two-stroke-cycle engine with a piston charging pump the lag of the charging pump must be adapted to these conditions, i.e., the angle of lag ψ with respect to the working crank must be so chosen that the maximum pressure of the pump, required for the use of the scavenging medium in question, will occur 20 to 30° before the lower dead center of the working piston. If, for example, the working piston has a total scavenging crank angle of $2\,\psi_s = 120°$ and if the useful compression stroke of the charging pump is likewise 120°, the timing is accordingly correct, when the angle of lag ψ is so chosen that a pump pressure of 0.45 atm. is produced about 25° before the bottom dead center with a consumption of the scavenging medium of $\lambda_s = 0.8$ at $n = 3{,}000$ r.p.m. In the most common case, the scavenging time cross sections of the engine and of the pump must be changed. Of course the same timing is suitable for other devices for supplying the scavenging medium, such as rotary and centrifugal compressors, blowers, etc. The prevalent idea that the pressure wave comes from the scavenging pump at the moment of opening of the scavenging ports (reference 4) proceeds therefore from fundamentally false assumptions regarding the expansion of the exhaust gases.

As already mentioned, the scavenging-type coefficient β plays an important role. The theoretical scavenging pattern attainable with the best value of β for the cross scavenging is represented in figure 5. The maximum two-dimensional scavenging-type efficiency for this longitudinal cylinder section is not smaller, as one would expect from practical experience (in accordance with which the loop scavenging actually yields better results), but even larger than that of the loop scavenging (fig. 3) and indeed because the scavenging port at the same dead center (upper edge of exhaust port) can be higher than in the cross-scavenging engine. The latter engine must therefore be capable of being brought, at the same power, to an equivalent specific fuel consumption, provided it is possible to attain the scavenging pattern represented in figure 5. Obviously this is impossible with the ordinary deflecting piston. The scavenging pattern is really very unsymmetrical toward the exhaust side. The vortex pocket 7 is smaller, but this is offset by the large exhaust vortices d_1 and d_2 in the cylinder head. It is known that, even in the cross-scavenging type, the difficulty consists in deflecting the scavenging current 1, 2, 3, as shown in figure 5. This is the only difference as compared with the loop scavenging. In the idealized pattern, the cross scavenging is even better. The height of the scavenging port should be ample, so as to yield the greatest possible value of η_{sf}. According to previous investigations, the size is correct, when $l_s = \sigma' s =$ 0.20 to 0.25 D.

Certain two-stroke-cycle engines have the following characteristics:

V_h cm³	s (mm)	D (mm)	s/D	$\beta = \frac{\sigma' s}{D}$
200	63	64	1.01	0.190
	72	59	1.22	0.204
	70	60	1.16	0.192
300	68	74	0.92	0.175
	72	72	1.0	0.166
	82.5	68	1.21	0.199

One is accordingly tempted to lay down the principle that two-stroke-cycle engines should have a large stroke-bore ratio. This requirement is also advocated by Lutz. Experience shows, however, especially in the case of high-speed engines, that this seemingly obvious conclusion is not universally valid, since elasticity, high speed, and low speed all require more or less opposite measures in such engines. The correct balance between the two requirements must rather be found for each individual case. Under all circumstances, stroke ratios below $s/D = 1.0$ should be avoided.

Regarding item 2, the quantitative dimensions of the scavenging ports are determined:

a) From the amount λ_s of the scavenging medium used,

b) From the scavenging pressure p_s,

c) From the maximum revolution speed,

d) From the expansion process in the working cylinder.

The scale factor is determined as the time cross section Z_s (m^2 s, $cm^2 \times 5$) or as the angular cross section W_s (m^2 deg., cm^2 deg.). Strictly speaking, the expansion process is unknown in the designing. Since, in all high-speed engines, this extends far into the geometric scavenging period, the mean scavenging time cross section must be 20 percent greater than would follow from a) and b).

For the quantitative dimensioning, it does not matter at first whether the calculated scavenging time cross section is obtained through high and narrow or low and wide scavenging ports. According to the preceding, such ports are to be regarded as high when the value β is greater than 0.20 to 0.25, while ports whose scavenging-type coefficient is much smaller than 0.20 must be regarded as low. If given the choice, one would be tempted to prefer the low ports, in order to lengthen the useful stroke and also to prevent a too long and violent injection of the hot exhaust gases into the scavenging port.

The strictly scavenging viewpoint does not admit of this choice, since it requires exactly the opposite. The often necessary and very costly testing on the stand is saved by making the scavenging ports at the outset as high as is compatible with the other requirements. Without exception it has always been found that the scavenging ports, contrary to the tendency in other kinds of construction, must be made higher than projected.

The adaptability of the ports is important. It is obvious that these difficulties would be eliminated by liners, especially when the ports are distributed over the whole circumference. Unfortunately no beginning has yet been made in this respect in high-speed two-stroke-cycle engines.

It has been shown that the quantitative dimensioning of the scavenging ports is correct, when the specific scavenging time cross section (referred to one cubic meter piston displacement) in high-speed engines at 3,000 r.p.m. with a scavenging efficiency λ_s ($= V_s/V_h$) of 0.65 is about $(Z_s)_s = 90$ cm^2 s/m^3 or the specific angular cross section $(W_s)_s = 1.6 \times 10^6$ cm^2 $deg./m^3$. For engines requiring more of the scavenging medium, the given number must be multiplied by the quotient λ_s'/λ_s. For engines coming under practical consideration, maximum values of about 200 cm^2 s/m^3 at 3,000 r.p.m. are thus obtained.

The factors λ_s, p_s, n, and p_z, which determine the size of the scavenging port, mutually depend on one another.

Such a high scavenging pressure is not practical for the following reasons:

a) The energy N_p required to supply the scavenging medium is excessive;

b) The final compression pressure p_c and temperature t_c in the working cylinder is too high;

c) The scavenging efficiency $(\eta_s, \eta_s', \eta_{sf})$ is poor.

The magnitude of λ_s depends especially on b and c. The scavenging mechanism (piston pump, centrifugal pump, rotary blower, etc.) compresses hourly V_s 60 n (m^3 : [p_o, T_o]) from the atmospheric pressure to the scavenging pressure p_s. The compression is first made at the pressure p_i, resulting from the compression ratio and the valve setting of the pump and then, with constantly increasing revolution speed, at the higher pressure p_s before the scavenging ports. The isothermal power for this is

$$N_{is} = C\ \lambda_s\ \log p_s/p_o\ \text{(hp.)} \tag{5}$$

where $V_s = \lambda_s V_h$ and $C = 0.0852\ p_o\ V_h\ 60\ n$

With λ_s remaining constant, N_{is} increases as represented in figure 6, which shows the characteristics of a high-power two-stroke-cycle engine of one liter piston displacement, such as may perhaps be realized within a few years. The conditions are made as favorable as permissible and a scavenging pump is assumed in which $\lambda_s = 1.0$ remains constant throughout the revolution-speed range. With the mean effective pressure $p_e = 7$ kg/cm² (99.56 lb./sq.in.), the power at 3,500 r.p.m. (n) is about 54 hp. The figure shows the full-load characteristic plotted against the scavenging pressure p_s, which increases with the speed. (For the relation between the abscissa values and the vertically plotted revolution speeds, see figure 9.) It is shown that the percentile share ψ of the power N_p of the scavenging pump increases greatly, because the isothermal efficiency η_{is} is relatively poor at all densities and seldom exceeds 0.65, while generally remaining between 0.45 and 0.55. According to the type of the compressor, the share ψ, with respect to N_e, can (through air turbulence, mechanical losses, etc.) reach 25 percent and more.* The statements of $\psi = 5$ to 8 percent for N_p, often found in technical reports, are therefore never warranted. If λ_s is made still greater than in this example, ψ then acquires values which can no longer be accepted. Such engines cannot be operated economically.

Scavenging Pressure and Consumption of the Scavenging Medium

The scavenging pressure increases with the consumption λ_s of the scavenging medium according to the increase in the scavenging time cross section and the actual scavenging time. It depends therefore essentially on the height of the scavenging port.

The following values were obtained with a well-known 200 cm³ (12.20 cu.in.) crankcase engine in agreement with measurements on equivalent engines of other types:

Specific scavenging time cross section $(\int f_s\, dt)_s = (Z_s)_s = 95$ cm² s/m³ (3,000 n);

Percentile scavenging-port height $\sigma' = 0.187$;

Scavenging-medium consumption at 3,000 n = $\lambda_{s_1} = 0.65$;

Scavenging pressure $p_{s_1} = 0.35$ atm.

This engine can be run successively (after installation of its own scavenging pump) with ever greater consumption of the scavenging medium. The scavenging pressure corresponding to every λ_s can be approximately determined in advance. The result is shown in figure 7. Corresponding to the higher load and the requisite exhaust period, the pressure p_x in the exhaust passage in-

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*In present-day rotary compressors (centrifugal compressors, etc.), this is, in fact, the rule. Hence the N_p curve of figure 6 is now twice as high.

creases just enough, as shown in the figure.

The extended curve shows the course of the scavenging pressure. It shows that the velocity c_s of the scavenging medium at the ports must increase in proportion to the greater charge.

For the subcritical pressure zone

$$c_s = \varphi \sqrt{2g \frac{k}{k-1} R T_s \left[1 - \left(\frac{p_z}{p_s}\right)^{\frac{k-1}{k}}\right]} \tag{6}$$

If λ_{s1}, p_{s1}, p_{z1} and λ_{s2}, p_{s2}, p_{z2} are corresponding terms, we have

$$p_{s2} = p_{z2}\left(\frac{1}{1 - \left(\frac{\lambda_{s2}}{\lambda_{s1}}\right)^2 \left[1 - \left(\frac{p_{z1}}{p_{s1}}\right)^{\frac{k-1}{k}}\right]}\right)^{\frac{k}{k-1}} \text{ atm. abs.} \tag{7}$$

The temperature T_s is put constant, that is, it is correspondingly cooled for the greater consumption of the scavenging medium. The line shows that, even for $\lambda_s = 1.0$, the scavenging pressure must be 0.9 atm. Beyond $\lambda_s = 1.1$, where $p_s = 1.2$ atm., equation (7) is no longer valid, since the critical compression ratio $\beta = p_z/p_s = 0.53$ is exceeded. The curve, which (for reasons whose explanation would lead us too far) is not strictly accurate, agrees exceptionally well with measurements in the region $\lambda_s =$ 0.4 to 1.1.

Instead of equation (7), we can quickly and conveniently use the approximate formula

$$p_{s2} = \left(\frac{\lambda_{s2}}{\lambda_{s1}}\right)^2 p_{s1} \text{ atm.} \tag{8}$$

which is represented by the dash-dot line in figure 7. This is sufficiently accurate in practice. The figure shows that the scavenging pressure assumes impractically high values beyond $\lambda_s = 1.0$ and, further, that the normal port dimensions of the ordinary crankcase engines could suffice, at most, up to $\lambda_s = 0.9$. Of course the scavenging time cross section is actually increased with increasing λ_s. The only question is how far this is possible. It follows that adequate time cross sections can be found for every scavenging method with practically utilizable λ_s. Since interesting comparisons can be made between the individual types, the possible arrangements will be briefly discussed. In a superficial consideration, only one scavenging method can be employed in which the whole circumference of the cylinder can be provided with scavenging ports. This leads to the three following methods.

1. Use of the loop scavenging with exhaust and scavenging ports arranged one above the other (fundamental form I of the loop scavenging (M.A.N.));

2. Use of the unaflow scavenging from the bottom dead center to the top dead center or vice versa;

3. Use of the U-cylinder.

It is to be assumed that there is need of the largest possible time cross section only for a large consumption of the scavenging medium. In this case, which we are now considering, with arrangements 1 and 2, the outlets must be controlled. It is of no interest here as to how this is accomplished. In arrangement 3, the control is more or less effected by the unsymmetrical port control.

The height of the scavenging ports is determined by the scavenging pattern. Since $\int f\,dt$ increases only linearly with the width $\psi' D \pi$ of the ports but almost quadratically with the height $\sigma' s$, a long-stroke engine is required for large λ_s. With respect to the port bridges and the catching of the piston rings in the ports, the practically maximum value of ψ' is 0.60, which is used in the calculation.

If s designates the stroke, D the cylinder diameter (bore), $\lambda = r/l$ the connecting-rod ratio, and $2\,\varphi_s$ the total scavenging crank angle, we then have for the time cross section in cm² s (four-cornered ports)

$$\int_{-\varphi_s}^{+\varphi_s} f_s\,dt = Z_s$$

$$= \frac{30\,\psi'\,Ds}{n}\left[\left(2\sigma' - 1 + \frac{\lambda}{4}\right)\varphi_s + \sin\varphi_s - \frac{\lambda}{8}\sin 2\varphi_s\right] \qquad (9)$$

and for the scavenging-angle cross section (cm³ degrees)

$$\int_{-\varphi_s}^{+\varphi_s} f_s \, d\, p^o = W_s = 6\, n\, Z_s \tag{10}$$

Let us consider a cylinder of 250 cm³ (15.26 cu.in.) piston displacement, which has proved particularly efficacious. Represent the stroke ratio with respect to the inertia forces by $s/D = 1.5$. For the cylinder we then have $s = 90$ mm (3.54 in.) and $D = 60$ mm (2.36 in.). The results obtained with these values are given in the table, where σ' and ψ' refer to the scavenging ports and σ and ψ to the exhaust ports. We thus obtain the following comparison.

Unaflow Cylinder

For $\sigma' = 0.20$ the scavenging begins at the bottom dead center. The exhaust valve is in the cylinder head and opens 0.05 s before the opening of the scavenging port. The stroke loss is therefore $\sigma_v = 0.25$.

Loop-Scavenging Cylinder

Fundamental form I.- The exhaust and scavenging ports lie at an angular distance of δ from one another. With the stroke loss $\sigma_v = 0.25$, only the value 0.10 remains for σ'. Consequently the scavenging-pattern coefficient $\beta = \sigma' s/D$ reaches only the insufficient scavenging value 0.15. The attainable time cross section is only 0.357 times the Z_s of the unaflow cylinder. In order to attain a better β value, we must admit a stroke loss σ_v of at least 0.28.

Fundamental form II.- This designates the loop scavenging, in which the scavenging ports, lying at about the same height, are located beside the exhaust ports. One would think at first that the scavenging time cross section must be smaller than in the fundamental form I, because the latter has the whole cylinder circumference at its disposal. Such was found, however, not to be at all the case. With the same stroke loss σ_v, Z_s is not smaller, but over 40 percent larger. The scavenging form coefficient β is more favorable. The relations change rapidly when σ', of the fundamental form I, is made larger (column 2 of the loop-scavenging engine in the table, page 24).

U-Cylinder

The U-cylinder must, of course, divide the stroke volume. In the present case $D = 48$ mm (1.89 in.) and $s = 70$ mm (2.76 in.). The volume loss V_v (cm³) is the smallest among the possible arrangements. The time cross section is 0.62 times the Z_s of the unaflow engine. In the U-cylinder the connecting rod of the scavenging piston is articulated to the connecting rod of the exhaust piston. The simplified assumption is here made that the articulated connecting rod oscillates in a circle instead of in the actual ellipse.

Results

The unaflow cylinder yields the greatest time cross section; next comes the U-cylinder and then the loop-scavenging cylinder. The ratio reads 1 : 0.62 : 0.35 or 1 : 0.62 : 0.50. With a small stroke ratio, for example, $s/D = 1.03$ ($s = 70$ mm (2.76 in.)), $D = 68$ mm (2.68 in.), $V_h = 255$ cm³ (15.56 cu.in.), the unaflow cylinder (for $\sigma' = 0.20$ = constant) can always reach 150 cm² s/m³, that is, more than the loop-scavenging fundamental form I with the large stroke ratio 1.5 and almost as much as the fundamental form II. The unaflow cylinder can therefore be used at a higher speed than any other type. At 6,000 r.p.m., for example, the U-cylinder under consideration would have 106 cm² s/m³; the unaflow cylinder $Z_s = 75$ cm² s/m³ with $s/D = 1.03$. It is doubtless easier, however, to bring this single-cylinder engine to 6,000 n than the U-engine with opposed pistons.

The table shows that the types under consideration yield time cross sections which suffice for every practical use of the scavenging medium, if 3,500 to 4,000 r.p.m. is assumed to be the maximum. As already mentioned, the only question is the control of the exhaust ports by their own devices since, with the use of the scavenging medium of more than 1.0, the union of the charging blower with exhaust ports not automatically controlled would be purposeless and uneconomical. In this connection we will consider what maximum values of λ_s' are involved. The scavenging pressures are represented in figure 9.

Scavenging Pressure and Revolution Speed

In the two-stroke-cycle engine, the scavenging pressure increases with the revolution speed. The climbing speed depends on whether it is an engine with the amount of the scavenging medium dependent or not on the revolution speed. The former group includes those engines in which, due to the nature of the drive, the quantity of the scavenging medium generally differs for every revolution speed. The second group includes all engines in which the requisite quantity of the scavenging medium depends only on the volumetric efficiency of the scavenging pump. The first group is represented by an automobile engine, in which every position of the throttle valve generally corresponds to a different revolution speed. The second group includes the stationary engines.

In automobile engines, full load at any revolution speed occurs only in three exceptional cases, namely, on the test stand, in hill climbing and, lastly, at the maximum speed of the vehicle on level ground. In all other cases, which are far more numerous, the delivery of the scavenging medium is throttled. Since the scavenging pressure depends on λ_s, it is considerably less in driving than on the test stand, where the engine is generally operated at full load. Those conditions are represented in figure 8. The scavenging proceeds therefore more favorably in driving than on the test stand, and consequently the driving speed for the most favorable fuel consumption in the two-stroke-cycle engine is more pronounced than in the four-stroke-cycle engine. It is therefore favorable for the former, especially in hill climbing (where λ_s is large) not to open the throttle wide. In fact, it is in agreement with the main process that the fuel consumption (in liters per 100 km) of the two-stroke-cycle engine is generally much more favorable than that shown by the calculation on the basis of the b_e value (g/hpe/h) of the test report. On the basis of these simple considerations, the following claims can be made for an automobile equipped with a two-stroke-cycle engine: least possible gear shifting, most direct drive possible, still better high speed, because at moderate r.p.m. the scavenging pressure is low and consequently the scavenging efficiency and fuel consumption are also low. To be sure, the time cross section of the exhaust ports increases and also the possibility of greater loss of the scavenging medium but, since the velocity of the scavenging medium increases much faster with increasing scavenging pressure (at least in engines which use considerable quantities of the scavenging medium) the scavenging efficiency is nevertheless better than at higher revolution speeds.

Moreover, the increase in the scavenging pressure with the revolution speed is based on the following main grounds:

1. Lag of the expansion process in the working cylinder, due to decreasing exhaust time cross section;

2. Decrease of the scavenging time cross section;

3. Reduction of the scavenging and charging periods;

4. Increase in heat transfer to the scavenging medium (heating from exhaust gases, friction, air turbulence, etc.).

The scavenging pump compresses the scavenging medium from the pressure p_e (intake pressure 0.92 to 0.98 abs. atm. with open valve), corresponding to the position of the throttle valve, with the compression ratio δ, to the final compression pressure p_i according to the formula

$$p_i = p_e \cdot \left(\frac{1}{\delta}\right)^k \tag{11}$$

This theoretical initial scavenging pressure is valid only at high revolution speeds, because the scavenging medium at low revolution speeds expands in the scavenging ports and in part escapes from the pressure side even during the compression. Thus two-stroke-cycle engines with p_i = 1.3 and 1.6 abs. atm. show, even at 1,000 r.p.m., the same vanishingly low scavenging pressure of 0.10 to 0.15 atm. before the ports. The pressure p_i becomes important above a certain revolution speed, from which point p_s increases rapidly. The following formula has been verified for the increase in the scavenging pressure:

$$p_s = p_i + 0.40 \times 10^{-7} \lambda_s^2 n^2 \tag{12}$$

Figure 9 shows the results for different quantities of the scavenging medium independent of the revolution speed.

The theoretical final compression pressure p_1 is intersected by the ordinate passing through the light-running revolution speed n_l. Between n_l and $n = 1,500$ r.p.m. the scavenging pressure cannot be followed. In this zone it is almost independent of both n and λ_s. Behind this zone it increases in engines with correctly dimensioned ports, as shown in the figure, according to the square of the revolution speed and of the quantity of the scavenging medium. Figure 9 shows that a scavenging pressure of 1 atmosphere is to be expected in an engine running with $\lambda_s = 1.2$ at 3,500 r.p.m. Even at $\lambda_s = 1.0$, that is, with the volume of the scavenging medium equal to the piston displacement, p_s is still 0.75 atm. at this r.p.m., but is only 0.52 atm. at 2,500 r.p.m.

Pressures above 0.75 atm. are highly questionable from both the scavenging and thermal viewpoints. The specific power (output per liter) of high-speed two-stroke-cycle engines is limited by the qualitative and quantitative scavenging process, either with respect to the r.p.m. or to the quantity of the scavenging medium. The working ranges of charged and supercharged engines are thus characterized. The term "charged engine" here means an engine in which the quantity of the scavenging medium is so great that, after completion of the scavenging, there still remains in the cylinder a quantity V_n of the scavenging medium corresponding to the piston displacement. Since the charging efficiency $\eta_1' = \eta_s \lambda_s$, the value of λ_s must be greater than 1.0, because η_s is always less than 1.0. If η_1' is greater than 1.0, it is then a supercharged engine.

In this state of affairs, there is renewed reason for calling attention to the fact that the best endeavors should be devoted not merely to the quantitative but also to the qualitative improvement of the two-stroke-cycle engine. This necessitates still further intensive research in the fields of scavenging, carburetion, combustion, and the designing of practical fuel-injection pumps and new engine types.

The p_s curves in figure 9 are based on the assumption of a constant quantity of the scavenging medium. In reality this varies according to the diminishing volumetric efficiency η_1 of the scavenging pump with increasing revolution speed. If, for example, $\lambda_s = 1.2$ at 1,500 n and drops to 1.0 at 2,600 n and to 0.65 at 4,000 n, the actual scavenging-pressure curve for this engine can be plotted by joining the corresponding points of the individual p_s curves in figure 9. Figure 10 shows such an example. The scavenging-pressure curve p_s, due to the diminishing volumetric efficiency η_1, takes a turn at 2,000 r.p.m. and continually recedes farther from the steeply ascending curve $(p_s)\ \lambda_s$ = constant. The dash-dot line represents the above-mentioned phenomenon, where pi (in the present instance 0.3 atm.) is not valid. Such p_s curves with turning point are, in fact, frequently found in technical reports. In stationary engines, where the revolution speed increases less (say from 1,000 to 1,500 n), only the descending portion of the curve, sometimes plotted as a straight line, is often to be seen, although the tendency of the curve is quite different, as has been demonstrated. Moreover, the location of the turning point permits a sure conclusion regarding the processes in the scavenging pump and indicates when and how much the volumetric efficiency diminishes, and to what extent thermal peculiarities or flow phenomena can be removed.

Scavenging Considered Three-Dimensionally

As already explained, the result of the scavenging has hitherto been evaluated simply according to the form of the scavenging pattern in the largest longitudinal cylinder cross section, the axial section or, as here designated, the main longitudinal cylinder cross section. It is thus tacitly assumed that the scavenging in the neighboring longitudinal cylinder sections is effected in the same or in a very similar manner. Actually such is not at all the case. If the scavenging process is considered three-dimensionally, just as it really takes place, important conclusions are reached. The two-dimensional scavenging-type efficiency η_{sf} has already been designated as the ratio of the area swept by the scavenging current to the whole area of the longitudinal cylinder section involved. The three-dimensional consideration leads to the three-dimensional scavenging-type efficiency η_{sv} and therefore denotes the ratio of the actually scavenged portion of the cylinder volume (including the combustion space) to the whole cylinder volume.

On the basis of mathematical considerations the value

η_{sv} can be as little determined as one of the η_{sf} -values. Through fundamental considerations, however, we can find the measures which lead to the highest possible value of η_{sv}. These measures are valid for all scavenging methods, in which the scavenging medium, after reaching a maximum point in the cylinder, flows in the opposite direction. These scavenging methods are mainly cross-scavenging and loop-scavenging.

It has already been explained that the η_{sf} of any desired longitudinal cylinder section can be determined from the following factors:

a) Cylinder diameter,
b) Stroke ratio s/D,
c) Height of scavenging port $\sigma' s$,
d) Scavenging pressure p_s,
e) Shape of combustion chamber,
f) Shape of piston.

The factors a to d have already been discussed, while the even better-known factors e and f, have yet to be investigated.

In a two-stroke-cycle engine of bore D and stroke s, when an advance is made from the middle by the amount x on the X-axis perpendicular to the longitudinal cylinder axis (Z-axis), the stroke ratio changes from s/D to $s/D_x = s/2\sqrt{\frac{D^2}{4} - x^2}$, a function which is represented in figure 11. If several scavenging ports of constant height $\sigma'.s$ are arranged on the cylinder circumference, the scavenging-type coefficient β then changes from $\sigma' s/D$ to $\sigma' s/2\sqrt{\frac{D^2}{4} - x^2}$.

Accordingly, if, in a two-stroke-cycle engine, the scavenging ports on a certain portion of the cylinder circumference are all made of the same height, as has always been the case, the scavenging-type coefficient β, as calculated from the plane of the main longitudinal cylinder section (Y-axis), changes continually in the direction of the β-curve of figure 11. When, therefore, the best value of η_{sf} is realized in the principal plane by suitably dimensioning the scavenging ports, it must diminish considerably in all other planes X_1X_1, X_2X_2 passing through the adjacent scavenging ports. The course of the scavenging with wide-open scavenging port (where η_{sf} is a maximum) is represented in figures 12 to 15 for the cross scavenging of the ordinary type. Figure 15 shows a cylinder cross section which corresponds to the five scavenging ports S_1 to S_5. Figure 13 shows the best flow in the longitudinal cylinder section X_1X_1 through the ports S_2 and S_3; figure 14, the flow in the section X_2X_2 through the ports S_4 and S_5. With respect to the no longer suitable values of the factors a to d in the sections X_1X_1 and X_2X_2, the scavenging current 1, 2, 3 in figure 13 turns prematurely, giving rise to the vortices 4, 5, and 6. The stroke ratio s/D_x is extreme in the longitudinal cylinder sections X_2X_2 supplied by the scavenging ports S_4 and S_5. The scavenging current, in order to reverse its direction of flow, would have to turn sharply at the cylinder head, which is of course, impossible. Hence it actually forms the short circuit shown in figure 2, and the scavenging current flows across the cylinder into the outlet passage ("cross-current scavenging").

The scavenging is as complete as possible when the maximum two-dimensional scavenging-type efficiency is attained in all planes passing through the individual scavenging ports. This assumes similar flow conditions. These are obtained by keeping the scavenging-type coefficient $\beta = \sigma' \, s/D_x$ constant. This is done by keeping the scavenging ports, corresponding to the variable value s/D_x, lower in proportion as they are farther removed from the XX-plane. In order that the scavenging currents, introduced through the scavenging ports which are always becoming lower, may actually rise to the cylinder head d, the scavenging pressure p_s is correspondingly raised, so that the scavenging pressure increases as the height of the ports diminishes.

The scavenging effected by these measures is represented by figures 16 and 17, which serve for comparison with the corresponding figures 13 and 14. The flow extends, moreover, to the cylinder head. Here the vortex pocket is greater, but the two-dimensional scavenging-type efficiency is nevertheless considerably higher, as the comparison very clearly shows. The designated arrangement is advantageous in automobile engines for attaining a good low-speed operation. It is equally suited for loop scavenging.

When the β coefficient in the customary arrangement does not indicate the requisite value in the main longitudinal cylinder section, β is better than in this section, due to the decreasing diameter D and consequently, η_{sf} in the longitudinal cylinder sections X_1X_1. In the cross sections still farther removed from the XX plane, η_{sf} must again decrease considerably. In this engine therefore the scavenging changes suddenly from poor to good and vice versa, resulting in a very unstable scavenging pattern. Since, however, the main longitudinal cylinder section multiplied by the width of the corresponding scavenging port S_1 represents the maximum part of the cylinder volume to be scavenged, even this engine is based on the above-described arrangement with constant β coefficient and scavenging pressure.

It is not possible to represent the existing relation between the qualitative and the quantitative scavenging efficiency on the one hand and the three-dimensional scavenging-type efficiency η_{sv} on the other hand. The higher η_{sv} is, just so much smaller are the leakage losses and just so much higher is η_s, as shown by figures 16 and 17. On the other hand, the greater η_{sv} is, just so much smaller is the share of the residual gases in the new charge and just so much higher is η_s'. The value η_{sv} furnishes a direct criterion for the result of the scavenging on the whole. Its experimental determination, however, is extremely problematical.

Eliminating the Vortex Pocket

The three-dimensional scavenging-type efficiency can be further improved, when one succeeds in driving out the vortex pocket 7 which is formed in all the scavenging methods, excepting the uniflow scavenging. In cross scavenging, this pocket is located immediately over the piston head. It is worth noting that the vortex pocket automatically increases, the more completely the scavenging current 1, 2, 3 flows through the cylinder. In the loop scavenging, fundamental form I, it lies between the inflowing and outflowing scavenging current, while in the fundamental form II it lies partly over the piston head and partly as in the fundamental form I. A vortex pocket, in the proper sense of the term, is formed especially in turbulent scavenging. Here the scavenging proceeds about a rotating pocket on the axis of the cylinder. This pocket is not driven out and impairs the three-dimensional scavenging-type efficiency.

What first concerns the elimination of the vortex pocket 7 (fig. 12) can be determined by the following simple device. The piston b (fig. 18) contains a trumpet-S-shaped passage f, of which one opening g, is directed toward the scavenging port S and the other opening h, is directed toward the inside of the cylinder. During the downstroke of the piston the scavenging current enters the cylinder a, according to its gradually developed excess pressure at the position of the piston shown in figure 18. The narrow scavenging current 1, 2, 3 first begins to flow as shown in this figure. This first phase of the scavenging does not last so long as the second phase (fig. 19). At the instant when the opening g, of the piston passage f, directed toward the scavenging port S in the further descent of the piston, is opposite the scavenging current 1, 2, 3, a portion of the scavenging medium flows through the S-shaped passage f and leaves by the opening g. In so doing, it partially expels the vortex pocket 7 and partially forces it into the path of the scavenging current 1, 2, 3, by which it is then expelled. During this process the scavenging current 1, 2, 3, first flowing according to figure 18, is simultaneously driven by the scavenging current 8, 9 from below toward the cylinder head d, and is thus prevented from forming as represented in figure 13 in the much longer second phase of scavenging. At the same time, the flow is improved. After a corresponding alteration of the piston, this arrangement can also be employed for the loop scavenging.

The elimination of the rotating vortex pocket 7 in the turbulent scavenging is most simply accomplished by means of a parallel current flowing along the axis of the cylinder inside the vortical current. In this process the parallel current must be admitted subsequently into the cylinder, in order to avoid loss of the scavenging medium. In the flow from the piston bottom dead center, as it occurs in the opposed-piston engine or in the U-cylinder, the parallel current is introduced through several piston passages e, corresponding to the vortical-current passages S and centrally united as shown in figure 20. The vortical-current passages, which supply the piston passages e, are lower than the others, so that the parallel current, which has a shorter distance to go, will be introduced correspondingly later. In the flow from the top dead center to the bottom dead center a valve serves to control the parallel current in the usual manner.

Summary

The viewpoints are discussed, according to which the scavenging of two-stroke-cycle engines can be evaluated, and the relations between the scavenging pressure and the quantity of the scavenging medium required, as also between the scavenging pressure and the revolution speed, are developed. It is further shown that the power increase is limited by the scavenging process, so that further researches are desirable for qualitative improvement. These results lead to several conclusions regarding the propulsion of motor vehicles by two-stroke-cycle engines.

Lastly, attention is called to the fundamental defect of the two-dimensional treatment of the scavenging process and to the consequent distinction between the two-dimensional and three-dimensional scavenging-type efficiency.

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National Advisory Committee
for Aeronautics.

TABLE

	Unaflow scaveng-ing	Loop scavenging Fundamental form I		Loop scaveng-ing Funda-mental form II	U-cyl-inder
s (mm)	90	90		90	70
D (mm)	60	60		60	48
s/D	1.5	1.5		1.5	1.46
V_h (cm^3)	255	255		255	254
σ	-	0.10	0.10	0.25	0.20
δ	-	0.05	0.05	0.05	-
σ'	0.20	0.10	0.13	0.20	0.20
σ_v	0.25	0.25	0.28	0.25	2X0.20
V_v (cm^3)	64	64	72	64	51
ψ'	-	0.60	0.60	0.30	0.60
2 φ_s deg.	116	82	94	116	116
$\beta = \sigma'$ s/D	0.30	0.15	0.195	0.30	0.292
Z_s (cm^2)$^{3000\,n}$	0.087	0.031	0.0455	0.0435	0.054
W_s (cm^2 deg.)	1570	588	820	785	973
$(Z_s)_s^{3000\,n}$	342	121	178	171	212
$(W_s)_s$	6.16×10^6	2.18×10^6	3.21×10^6	3.08×10^6	3.33×10^6
f_s (cm^2)	20.4	10.2	13.3	10.2	12.7
f_s' cm^2/cm^3	0.08	0.04	0.05	0.04	0.05

(The Z_s and $(Z_s)_s$ values are valid for 3,000 r.p.m.)

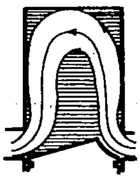


Fig. 1

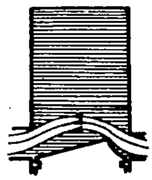
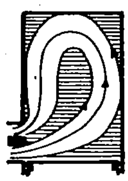


Fig. 2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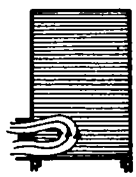
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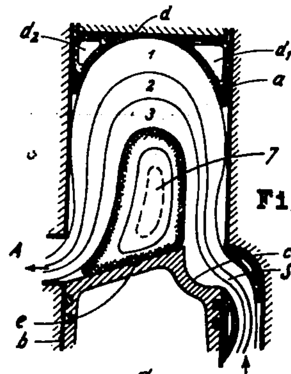


Fig. 5

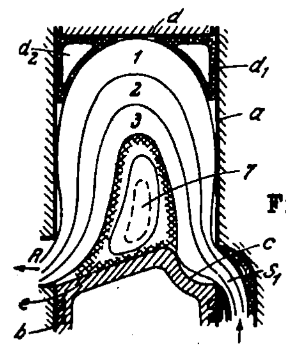
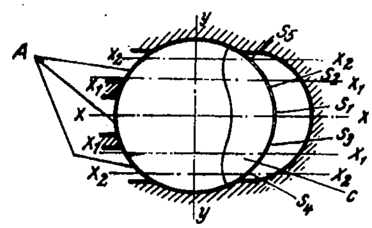


Fig. 12



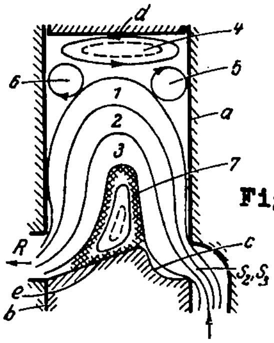


Fig. 13

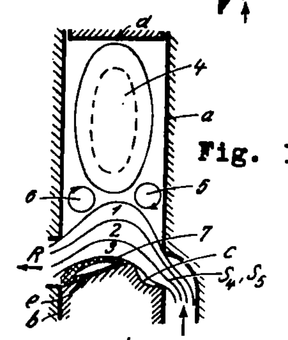


Fig. 14

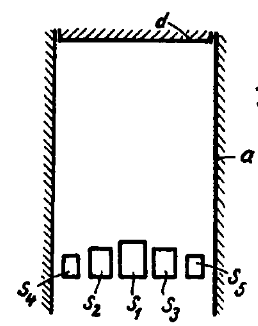


Fig. 15

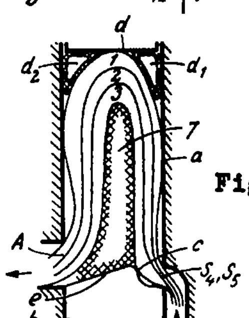


Fig. 17

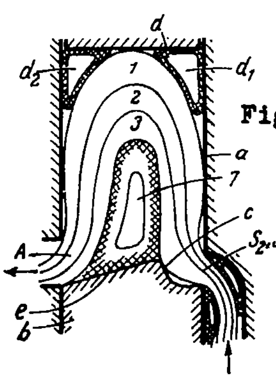


Fig. 16

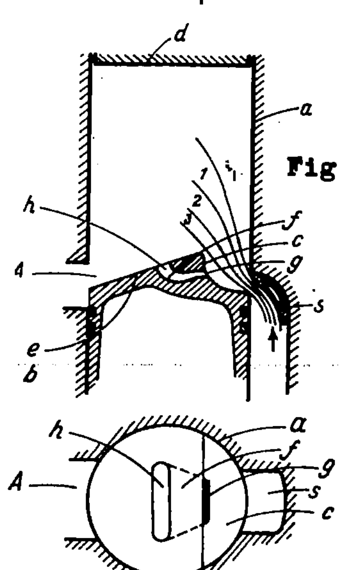


Fig. 18

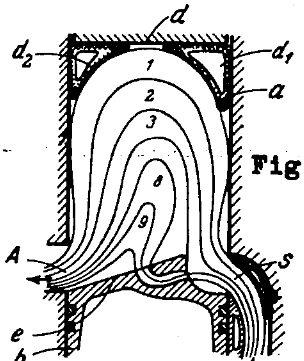
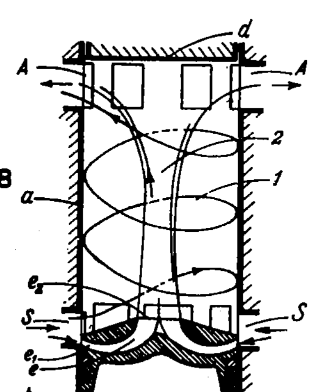


Fig. 19



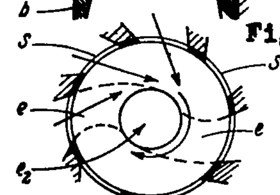
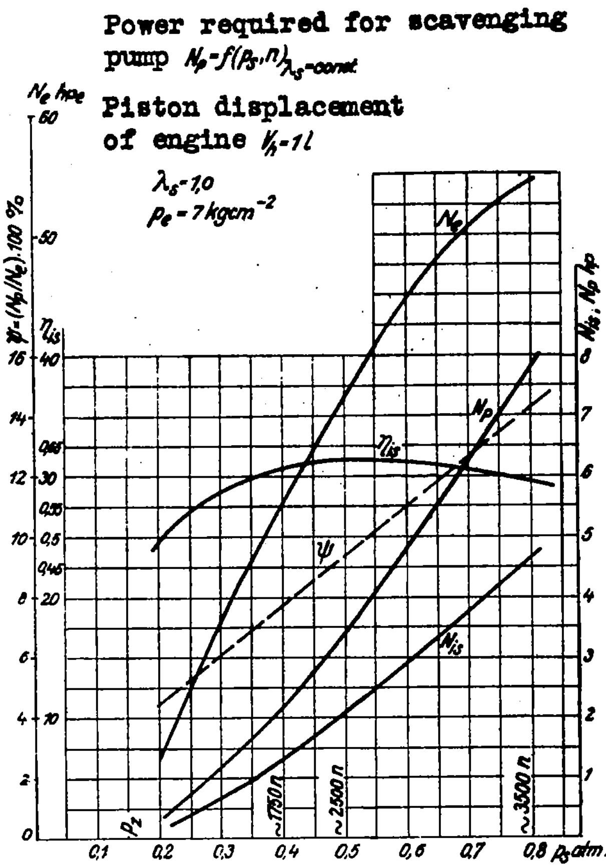


Fig. 20



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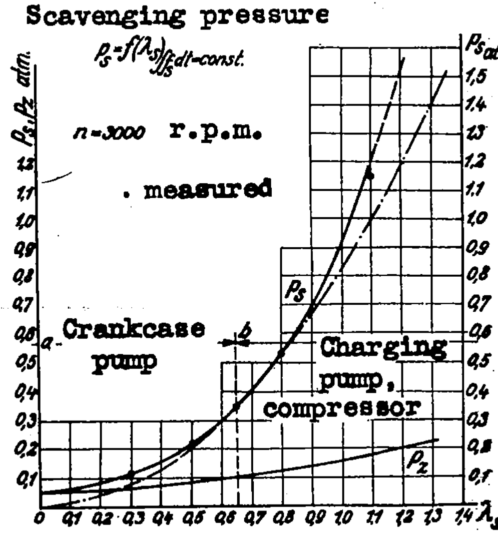

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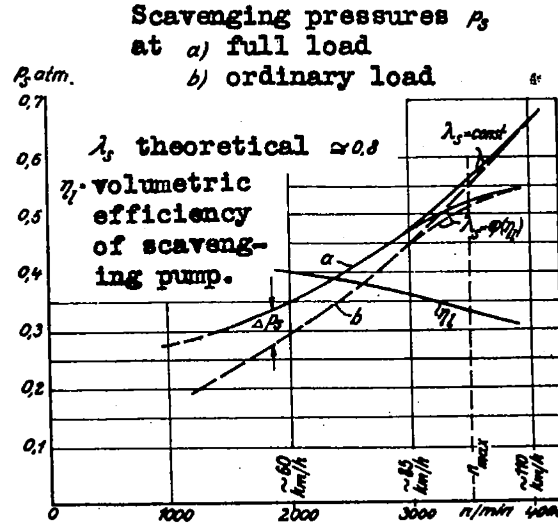

Figure 8

Scavenging pressure $p_s = C + f(n)_{\lambda_s = const}$

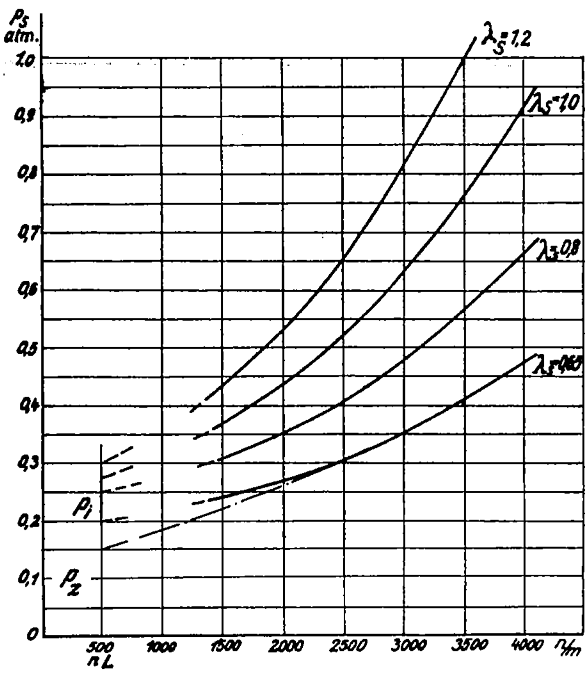

Figure 9

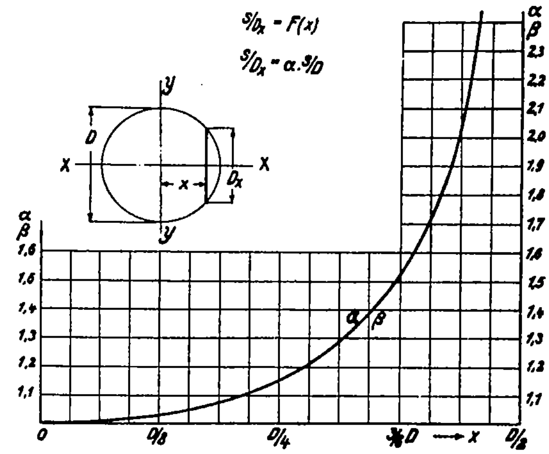

Figure 11

Scavenging pressure $p_s = f(\lambda_s, n)$

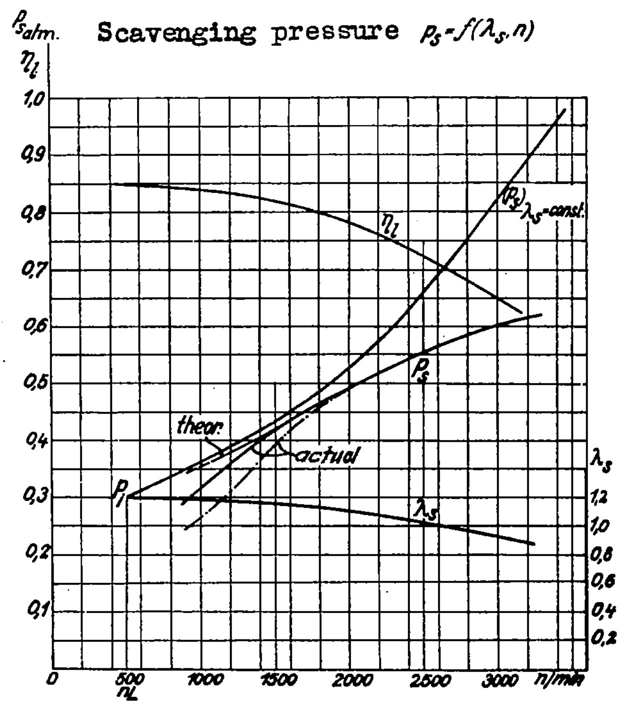

Figure 10

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